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**RADIAL INFLOW TURBINE PERFORMANCE WITH AN EXIT
DIFFUSER DESIGNED FOR LINEAR STATIC PRESSURE VARIATION**

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**CASE FILE
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This information is being published in preliminary form in order to expedite its early release.

ABSTRACT

Two diffusers were designed with the same area ratio but different variations between the inlet and exit. The first, with a cylindrical inner body and conical outer wall provided a rapid rise in static pressure at the inlet. The second diffuser was designed for a linear increase in static pressure from inlet to exit. Each diffuser was tested as part of the turbine with cold argon at design Reynolds number. Overall turbine total-to-total efficiency was increased from 0.894 to 0.905 in changing from the conical diffuser to the linear static pressure diffuser. The diffuser loss was reduced from 0.019 to 0.008 in terms of overall total-to-total turbine efficiency.

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SUMMARY

Two turbine exit diffusers were designed for the 2 to 10 kW Brayton Rotating Unit. The original had a cylindrical inner body and a conical outer wall. The second, designed with the same area ratio and slightly greater length, had a tapered inner body and an outer wall contoured to provide a linear increase in static pressure.

Both diffusers were operated as part of the turbine assembly and at design equivalent speed and Reynolds number. At design pressure ratio, equivalent speed, and Reynolds number, the second diffuser provided an overall turbine efficiency one point higher than did the original. This improvement indicates a potential increase of 2.8 percent in system electrical net power output.

Static pressure measurements and diffuser exit surveys indicate that the flow separated from the inner body causing a slight increase in velocity and hence slight reductions in diffuser efficiency and effectiveness. Diffuser performance might be further improved by inhibiting this separation.

INTRODUCTION

Turbomachinery components designed for Brayton cycle space power systems are being investigated to determine performance characteristics. Typically, a point in turbine performance is equivalent to about $2\frac{1}{2}$ percent in net system power output while a point in compressor efficiency is equivalent to about $1\frac{1}{2}$ percent. There is therefore considerable incentive to achieve maximum component efficiencies.

Components of the Brayton Rotating Unit (BRU) power generation system have been under investigation for several years at the Lewis Research Center. A 4, 97-inch diameter turbine drives the compressor and the alternator of this 2 to 10 kW system. The turbine assembly included a diffuser with a cylindrical inner body and a conical outer wall. The area ratio of the diffuser was 2.65. The turbine and diffuser designs are described in the contractor's report, reference 1. The turbine, with and without the diffuser, was tested in cold argon and the investigation was reported in reference 2. Results indicated that the diffuser loss was 0.02 in overall total-to-total turbine efficiency, 1.4 times the design loss.

Design flow characteristics in the diffuser were then examined with a one-dimensional calculation of velocity and static pressure as functions of flow area and total pressure loss. An alternate design was made in order to change the maximum deceleration rate and thereby reduce the diffusion loss. This diffuser was designed to provide a linear variation in static pressure through the diffuser. The rate of deceleration is much smaller near the inlet than in the original diffuser.

The second diffuser was built and tested as a part of the research turbine. This report describes the second diffuser design, its performance, and overall turbine performance. Design characteristics and performance of the original and the second diffusers are compared.

SYMBOLS

H	isentropic specific work (based on total pressure ratio) (ft-lb)/lb
Δh	specific work, Btu/lb
N	turbine speed, rpm
N_s	specific speed, $NQ^{1/2}/H^{3/4}$, rpm (ft) ^{3/4} /sec ^{1/2}
p	pressure, psia
Q	volume flow (based on exit conditions), ft ³ /sec
Re	Reynolds number $w/\mu r_+$
r	radius, ft
T	absolute temperature, °R

U	blade velocity, ft/sec
V	absolute gas velocity, ft/sec
V_j	ideal jet speed corresponding to total- to static-pressure ratio across turbine, ft/sec
W	relative gas velocity, ft/sec
w	weight flow, lb/sec
γ	ratio of specific heats
δ	ratio of inlet total pressure to U. S. standard sea-level pressure, p^t/p^*
ϵ	function of γ used in relating parameters to those using air inlet conditions at U. S. standard sea-level conditions, $(0.740/\gamma)(\gamma - 1/2)\gamma^{1/(\gamma-1)}$
η	turbine efficiency
η_s	static efficiency (based on inlet-total- to exit-static-pressure ratio)
η_t	total efficiency (based on inlet-total- to exit-total-pressure ratio)
θ_{cr}	squared ratio of critical velocity at turbine inlet to critical velocity at U. S. standard sea-level temperature, $(V_{cr}/V_{cr}^*)^2$
μ	gas viscosity, lb/(ft)(sec)
ν	blade-jet speed ratio (based on rotor-inlet tip speed) U_t/V_j

Subscripts:

cr	condition corresponding to Mach number of unity
eq	equivalent
t	tip
1	station at turbine inlet
2	station at stator exit

- 3 station at rotor exit
- 4 station at diffuser exit

Superscripts:

- ' absolute total state
- * U. S. standard sea-level conditions (temperature equal to 518.67° R, pressure equal to 14.70 psia)

TURBINE DESIGN

The 4.97-inch tip diameter radial-inflow turbine was designed for 6.0 kilowatt net electrical output with a xenon-helium mixture as the working fluid. The design-point values for the turbine are as follows:

Inlet total temperature, T_1' , °R	2060
Inlet total pressure, p_1' , psia	25
Mass flow, w , lb/sec	0.7484
Turbine rotative speed, N , rpm	36 000
Total- to total-pressure ratio	
Overall, p_1'/p_4'	1.749
Rotor exit, p_1'/p_3'	1.740
Total- to static-pressure ratio	
Overall, p_1'/p_4	1.763
Rotor exit, p_1'/p_3	1.800
Blade-jet speed ratio, ν	0.690
Total to total efficiency	
Overall, $\eta_{t, 1 \text{ to } 4}$	0.884
Rotor exit, $\eta_{t, 1 \text{ to } 3}$	0.897

Total to static efficiency	
Overall, $\eta_{s, 1 \text{ to } 4}$	0.875
Rotor exit, $\eta_{s, 1 \text{ to } 3}$	0.850
Specific work, Δh , Btu/lb	21.80
Reynolds number, $Re = w/\mu r$	76 200
Specific speed, $N_3 = NQ^{1/2}/(H)^{3/4}$, rpm (ft ^{3/4})/sec ^{1/2}	76

The following air equivalent (U. S. standard sea level) design values were computed:

Mass flow, $\epsilon w \sqrt{\theta_{cr}}/\delta$, lb/sec	0.4860
Specific work, $\Delta h/\theta_{cr}$, Btu/lb	14.82
Rotative speed, $N/\sqrt{\theta_{cr}}$, rpm	29 687
Total- to total-pressure ratio	
Overall, $(p'_1/p'_4)_{eq}$	1.658
Rotor exit, $(p'_1/p'_3)_{eq}$	1.645
Total- to static-pressure ratio	
Overall, $(p'_1/p'_4)_{eq}$	1.669
Rotor exit, $(p'_1/p'_3)_{eq}$	1.695
Blade-jet speed ratio, ν	0.690

Design velocity diagrams corresponding to these conditions and the selected inlet and exit diameters are shown in figure 1. Figure 2 shows a section through the research turbine with the original exit diffuser. The turbine tip diameter, as noted previously is 4.97 inch. Exit shroud and hub diameters are 3.480 and 1.822 inches, respectively.

DIFFUSER DESIGNS

The original turbine exit diffuser has inlet and exit areas of 6.890 and 18.290 square inches, respectively. The axial length measured from the rotor blade trailing edge is 9.754 inches. A section through the turbine and this diffuser is shown in figure 2. The conical outer wall provides an area variation with axial distance that is almost linear. A continuity calculation through this diffuser was made assuming that the design total pressure loss occurred linearly with distance from the inlet to the outlet. This calculation showed a rapid deceleration at the inlet and much slower deceleration near the exit. The associated inlet static pressure gradient was approximately twenty times as high as the exit gradient. The calculated values were 0.1 and 0.005 psi per inch.

The calculated pressures were then used to obtain a diffuser effectiveness value where effectiveness is defined as $(p_4 - p_3)/(p'_3 - p_3)$. The value corresponding to design total pressure loss is 0.661. The isentropic effectiveness is 0.856. Diffuser efficiency, defined as the ratio of actual to isentropic effectiveness, would therefore be 0.77 with design flow and design total pressure loss.

A second diffuser was designed with the same inlet and exit flow areas. A slightly greater length was permissible within the packaging envelope so the length was 11 percent greater than in the original. Static pressure was made linear with distance from the inlet in order to avoid rapid deceleration where the kinetic energy level was highest. The constant static pressure gradient was 0.029 psi per inch. The inner body of this diffuser tapered toward the exit as shown in figure 3.

The variation in total and static pressure through both diffusers is shown in figure 4. Linear total pressure losses are assumed as shown. The static pressures were calculated for uniform axial velocity; a one-dimensional calculation of continuity. The static pressure schedule is substantially different, particularly in the rate of change with distance near the inlet.

Figure 5 shows the area variation through each diffuser. The parabolic variation in the original diffuser is almost linear in this interval. The second diffuser, however, has a very gradual area increase near the inlet and then an increasingly rapid rate of increase toward the exit.

The velocities corresponding to the two previous figures are shown in figure 6. The difference, as noted previously, is pronounced.

EQUIPMENT AND OPERATION

The test rig and instrumentation are the same as those used in previous operation of the research turbine. A description of the flow system, dynamometer, data recording, etc., is included in reference 2.

The diffusers discussed in this report were tested only as parts of the turbine, never as individual components. Performance measurements therefore correspond to a diffuser inlet flow condition that includes high turbulence and radial gradients in flow, pressure, temperature, and flow angle.

Test operation with the second diffuser was run in cold argon with two inlet pressures, 7.0 psia and 20 psia. 7.0 psia with the test turbine inlet temperature of 610°R provides design Reynolds number. Evaluation of diffuser performance for this low pressure operation depends on small differences in measured pressures. Consequently, additional runs were made at 20 psia to substantiate the results obtained at 7 psia.

RESULTS AND DISCUSSION

A second turbine exit diffuser was designed in an effort to improve overall turbine performance. Diffuser and overall turbine performance were determined during operation over ranges of pressure ratio and speed. Test runs were made of two pressure levels; 7.0 psia in order to provide design Reynolds number, and 20 psia to increase the certainty of the pressure measurements. There was no noticeable difference in calculated performance determined from measurements at the two pressure levels. The performance curves in this report are from the design Reynolds number operation.

Overall Performance

Overall turbine total efficiency with the original diffuser is shown in figure 7 as a function of blade-jet speed ratio. This figure is taken from reference 2 (fig. 12(b)) and includes ranges of pressure ratio and speed. The solid line is faired through the design speed data points. The curve peaks at 0.894 near the design blade-jet speed ratio of 0.69. This is 0.01 higher than the design value of 0.884. The corresponding efficiency based on rotor exit total pressure was 0.913. The diffusion penalty was therefore 0.019 in total efficiency.

Figure 8 shows in the same manner performance with the second diffuser. The peak efficiency of 0.905 occurs at the design blade-jet speed ratio. The diffusion penalty indicated here is 0.008. This increase in overall turbine efficiency represents a potential 2.8 percent increase in net alternator power output. The design speed curves for both diffusers are in figure 9 for comparison. Table I summarizes diffuser and overall performance.

Static Pressure Distribution

Static pressures on the wall of the second diffuser were measured at five static tap locations between the rotor exit and the diffuser exit and one location downstream. Figure 10 shows pressures measured at design speed, pressure

ratio, and Reynolds number. Pressures are shown in ratio form because of the small differences involved and because flow controls did not provide perfectly steady operation. Scatter in individual readings was reduced by averaging the results of 18 data recordings. Experimental results agree closely with the design calculations, particularly near the diffuser inlet. The measured pressures drop below the design curves toward the exit, however. This apparently results from two effects. Diffuser effectiveness and efficiency were somewhat below the design values as shown in table I. This would increase exit velocities. Also, diffuser exit surveys showed that flow angle was not measurable at radii smaller than 1.047 inches. This region of no flow would also cause an increase in actual velocities by reducing the effective flow area.

The second diffuser showed an appreciable improvement in performance over the original diffuser. An additional improvement might result if flow separation from the inner body could be inhibited.

REFERENCES

1. Anon.: Design and Fabrication of the Brayton Cycle High Performance Turbine Research Package. Final Report. Rept APS-5281-R, AiResearch Manufacturing, Co. of Arizona (NASA CR-72478), 1968.
2. Nusbaum, William J. and Kofskey, Milton G.: Cold Performance Evaluation of 4.97-Inch Radial-Inflow Turbine Designed for Single-Shaft Brayton Cycle Space-Power System. NASA TN D-5090, 1969.

TABLE I. - DIFFUSER PERFORMANCE

	Diffuser effectiveness*	Diffuser efficiency*	Overall turbine efficiency
Design calculations	0.661	0.77	0.884
Experimental, original	.40	.47	.894
Experimental, second	.58	.65	.905

* See DIFFUSER DESIGNS for definitions.

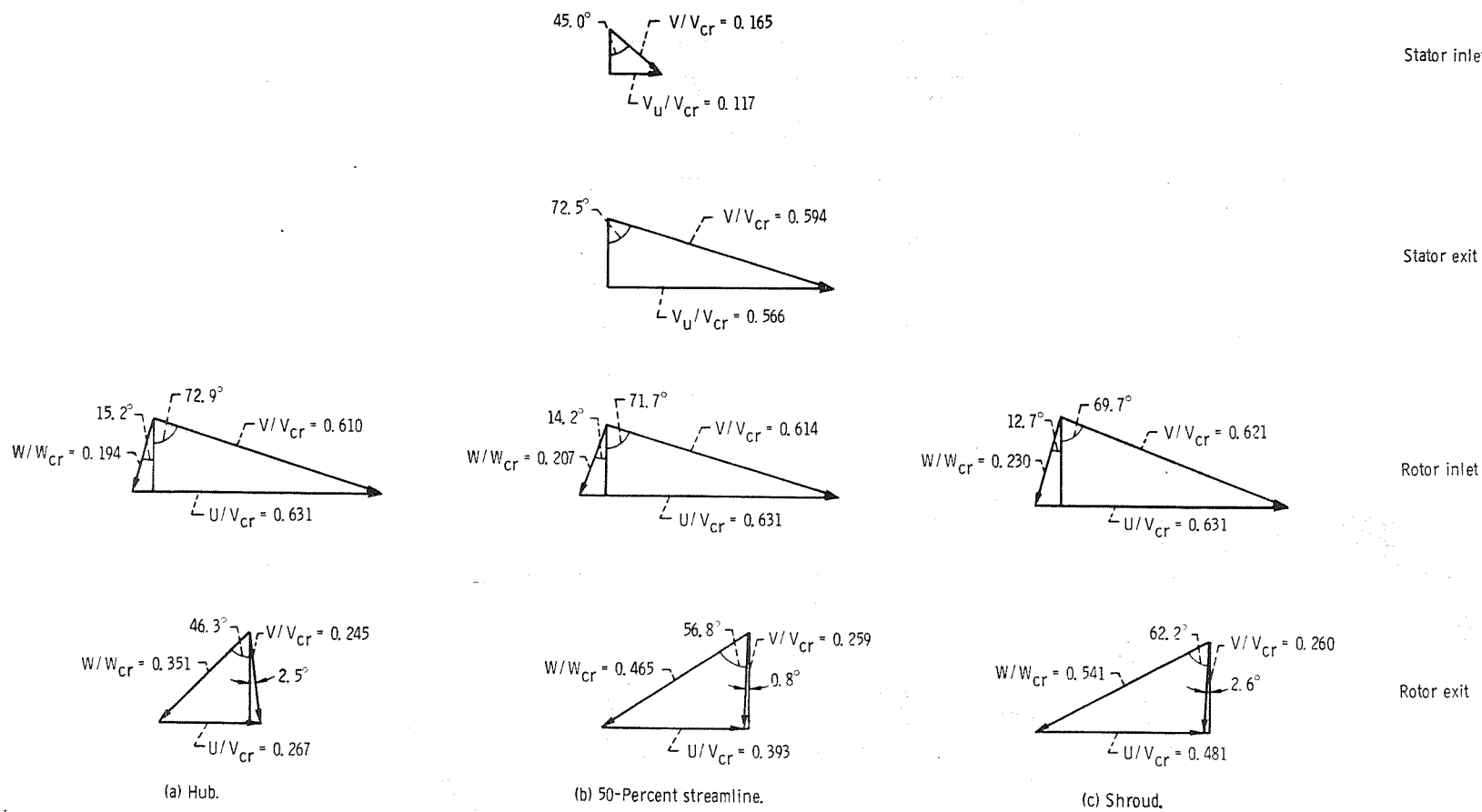


Figure 1. - Design velocity diagrams (inside blade row).

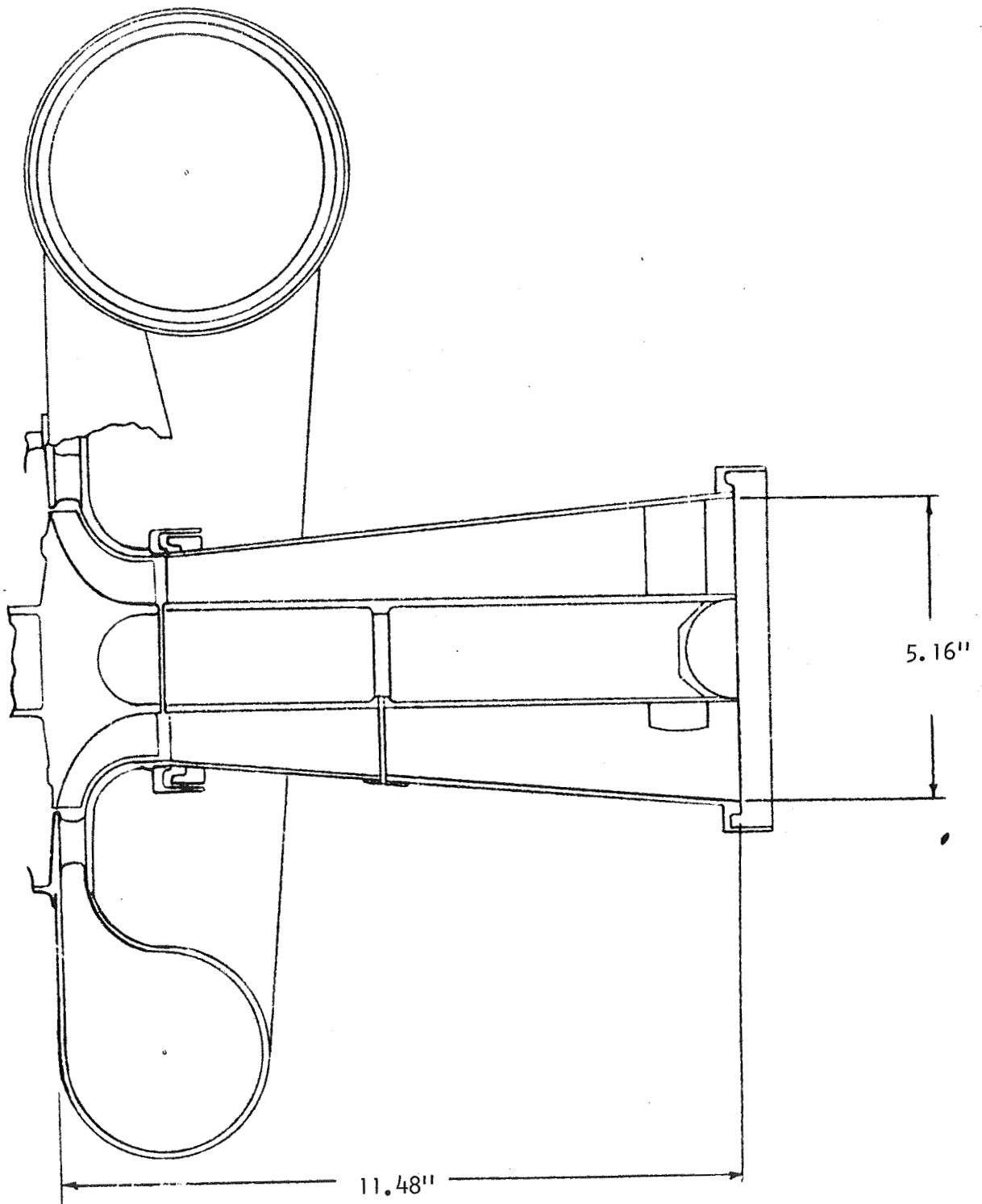


Figure 2 - Turbine with original diffuser.

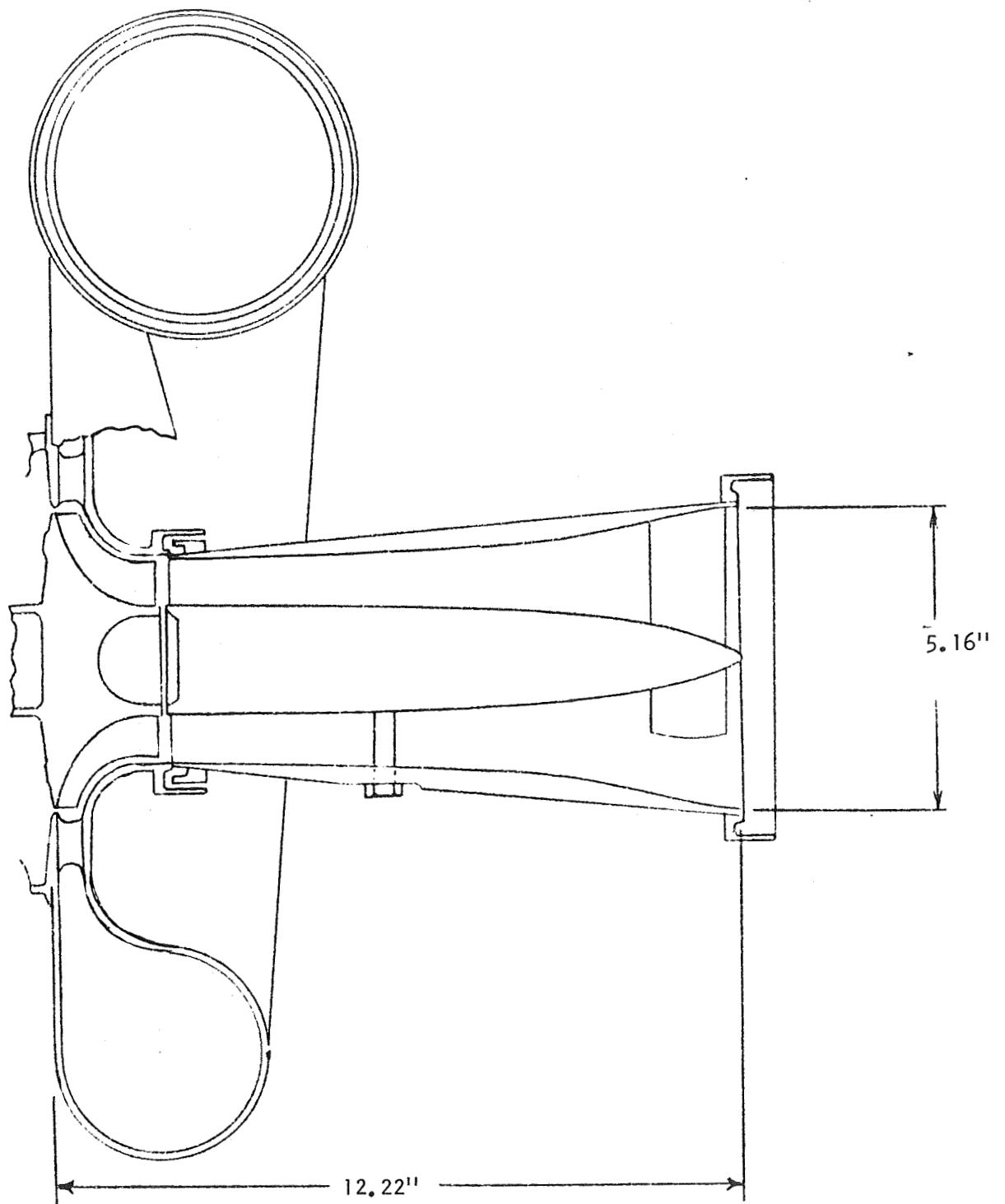


Figure 3 - Turbine with second diffuser.

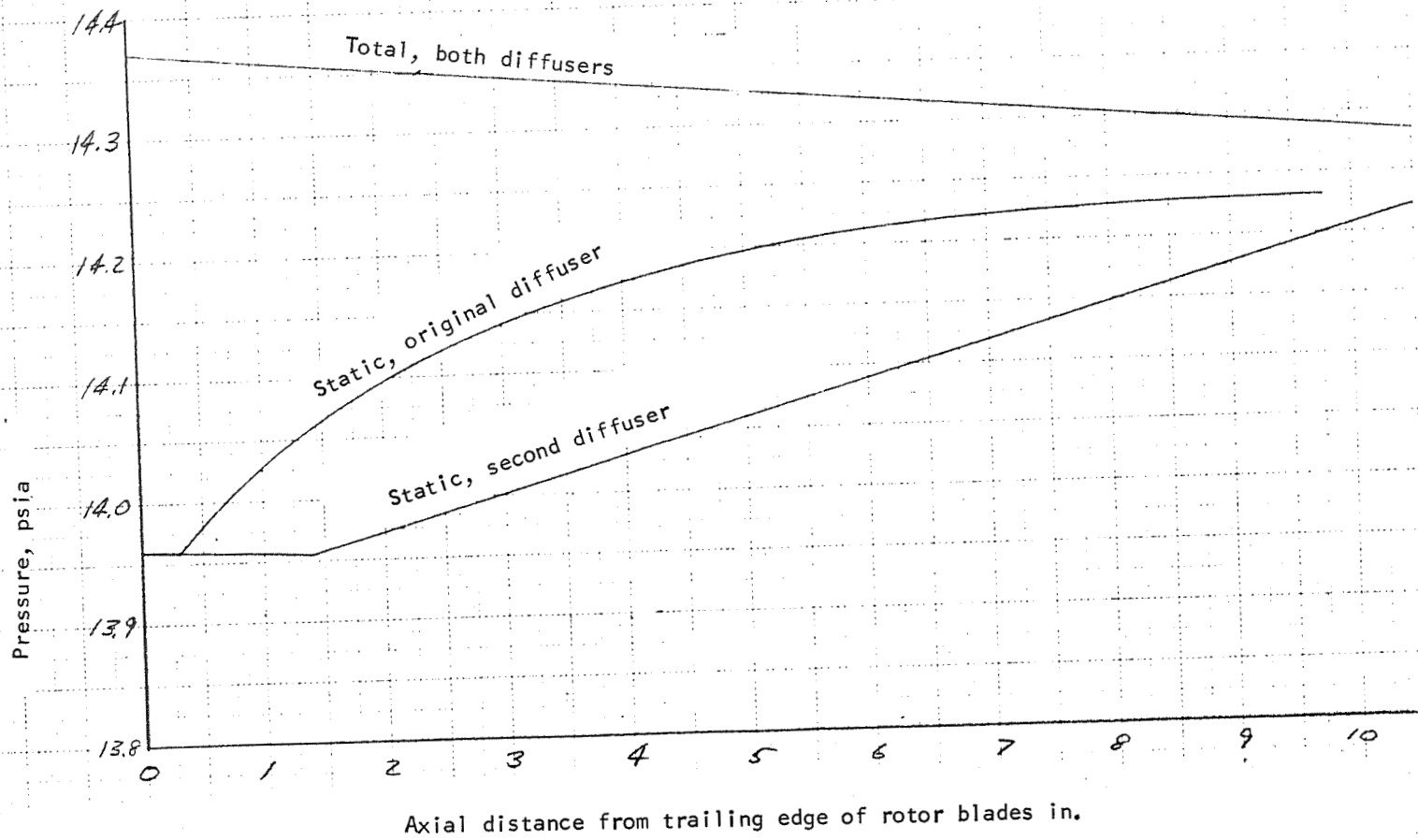


Figure 4 - Estimated pressure distributions in diffusers

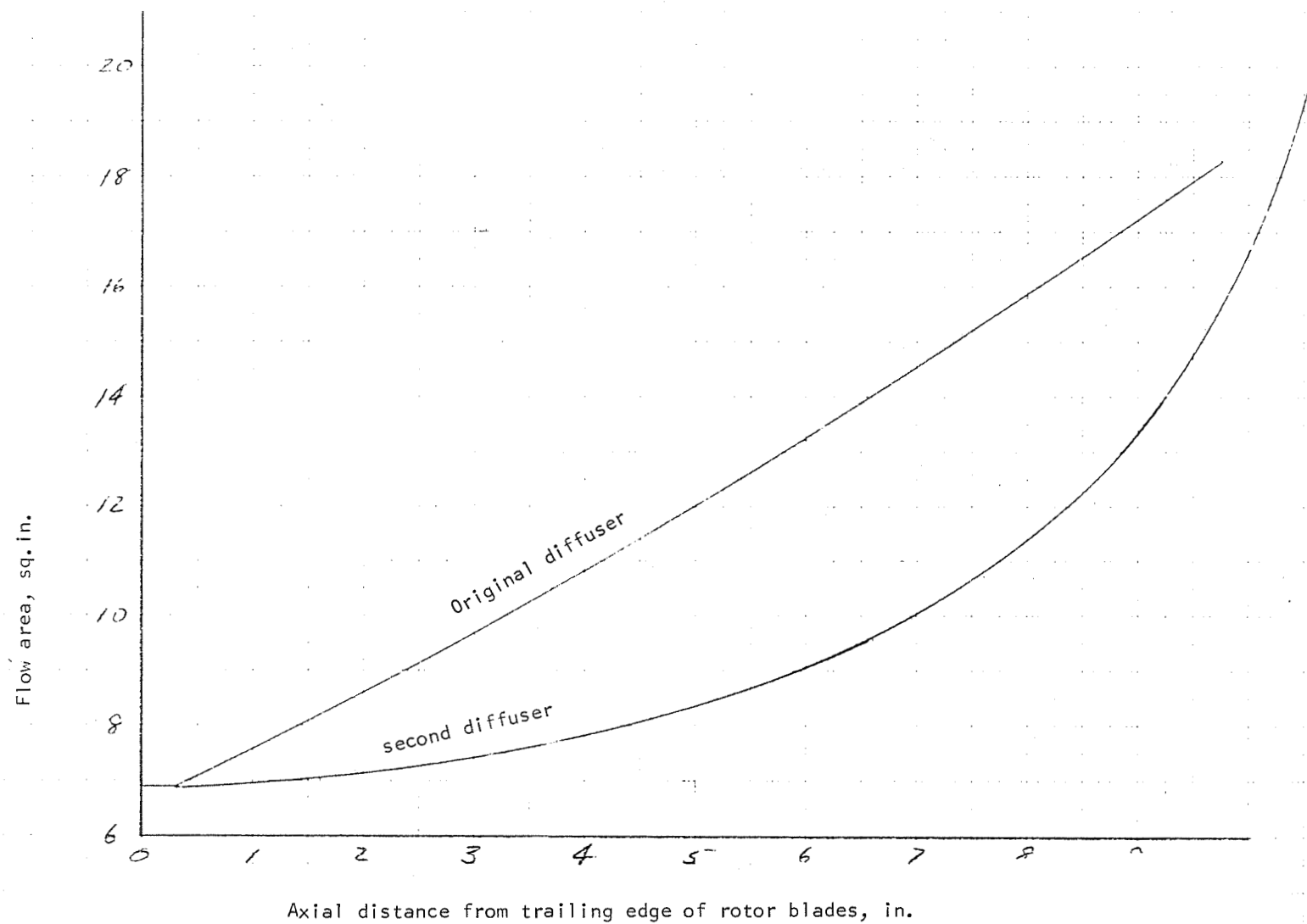


Figure 5 - Flow area variation through diffusers.

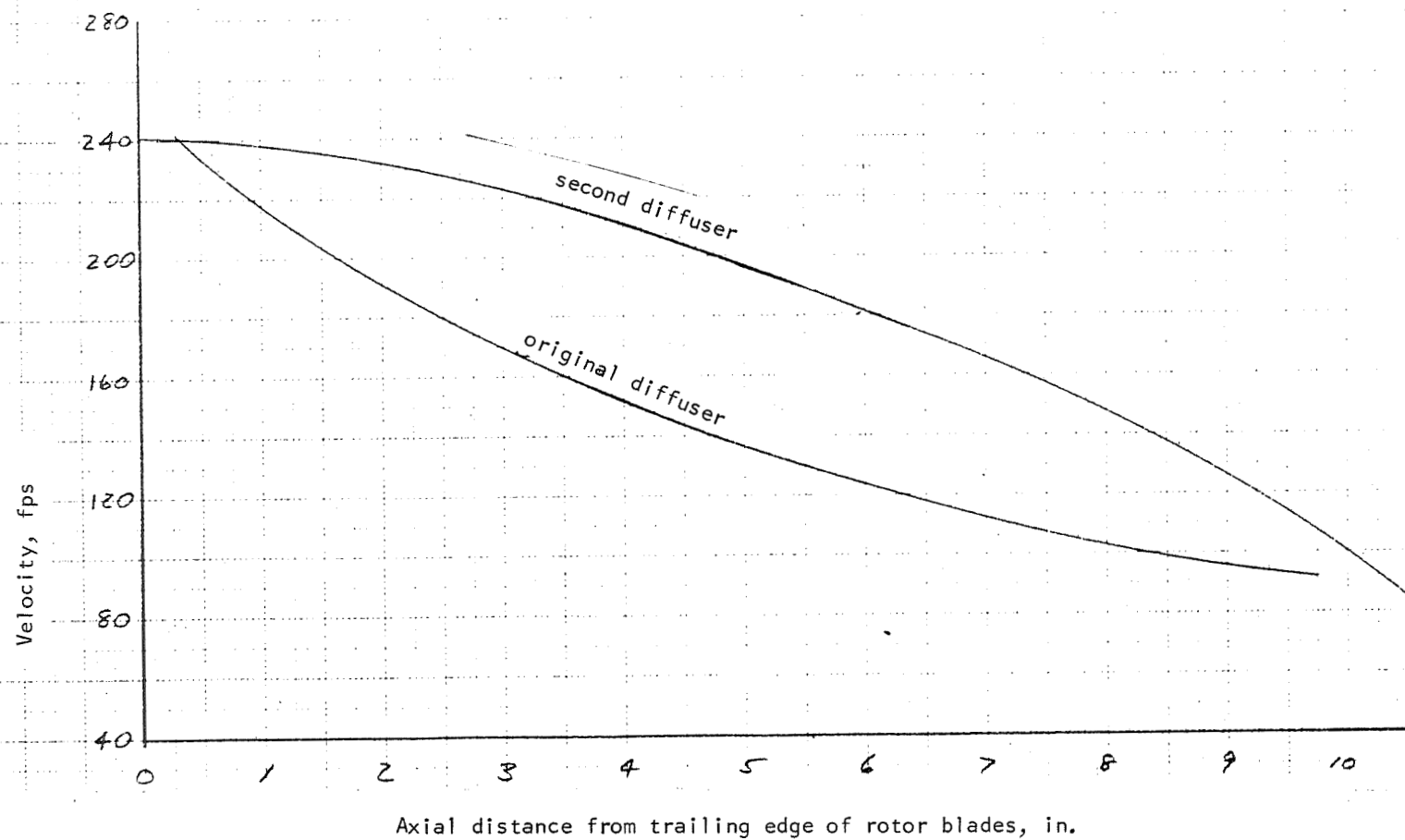


Figure 6 - Estimated velocity distributions through diffusers.

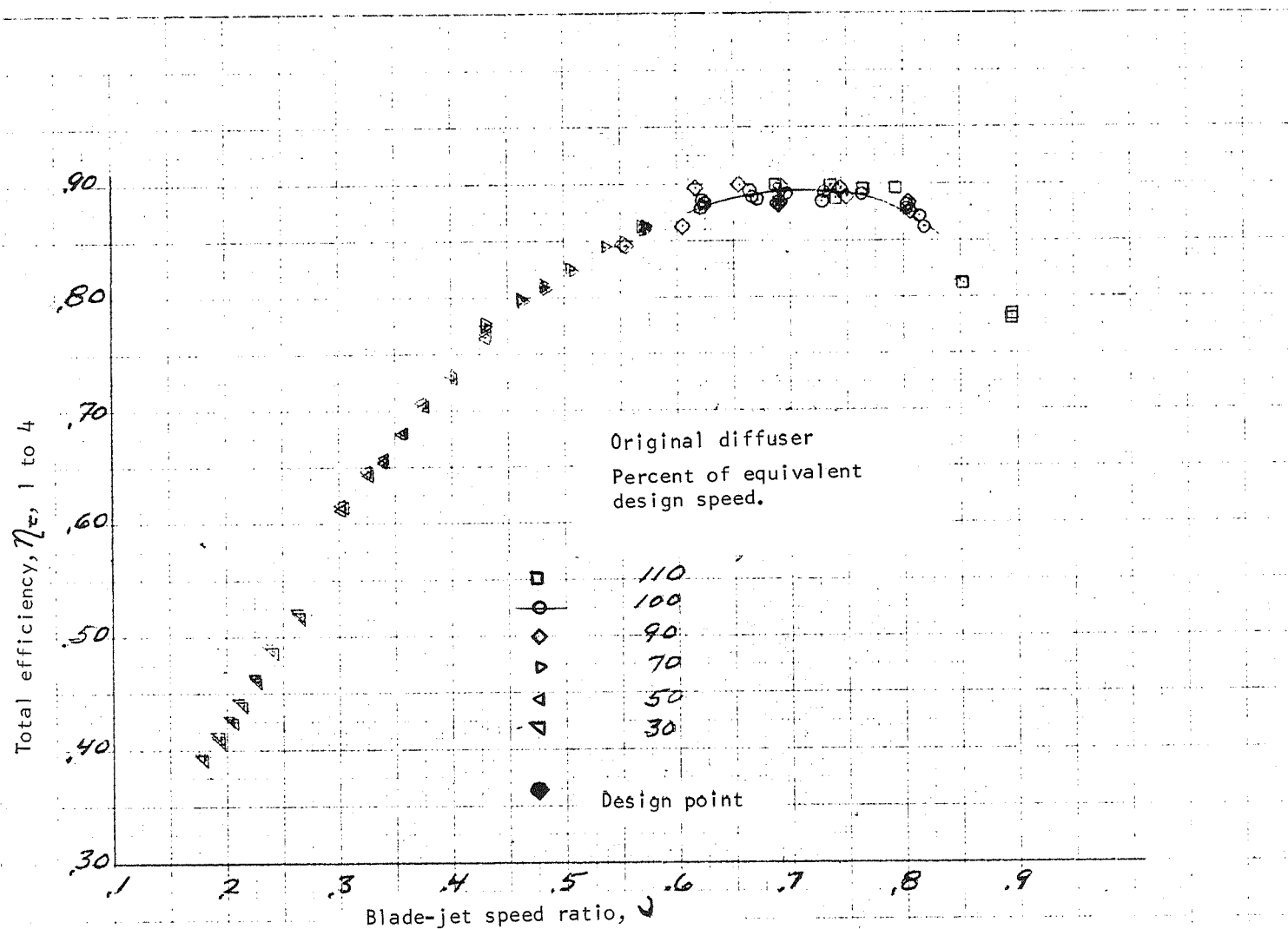


Figure 7 - Variation of total efficiency with blade-jet speed ratio. (where ideal work is calculated from conditions at the turbine inlet and diffuser exit)

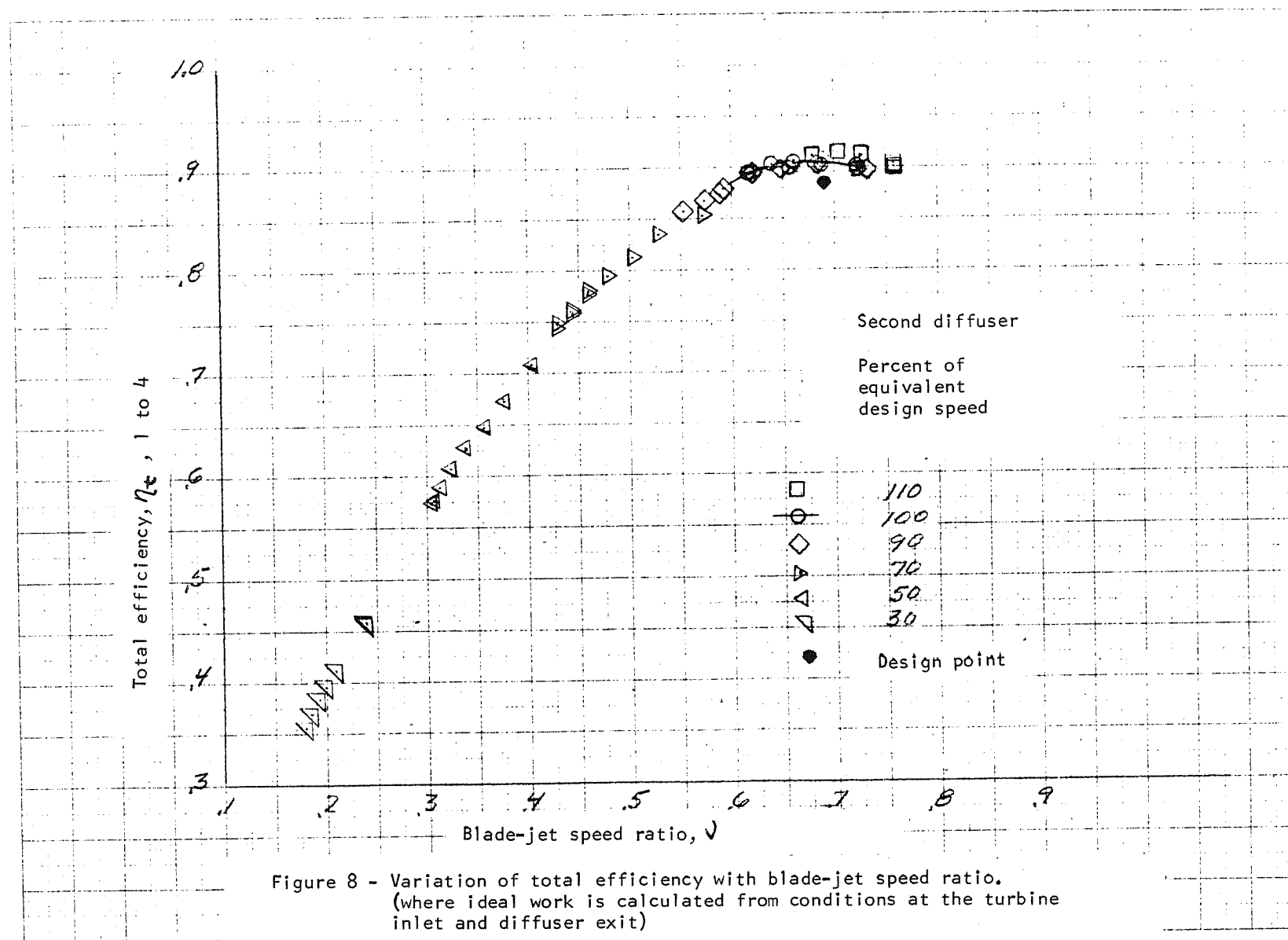


Figure 8 - Variation of total efficiency with blade-jet speed ratio.
(where ideal work is calculated from conditions at the turbine inlet and diffuser exit)

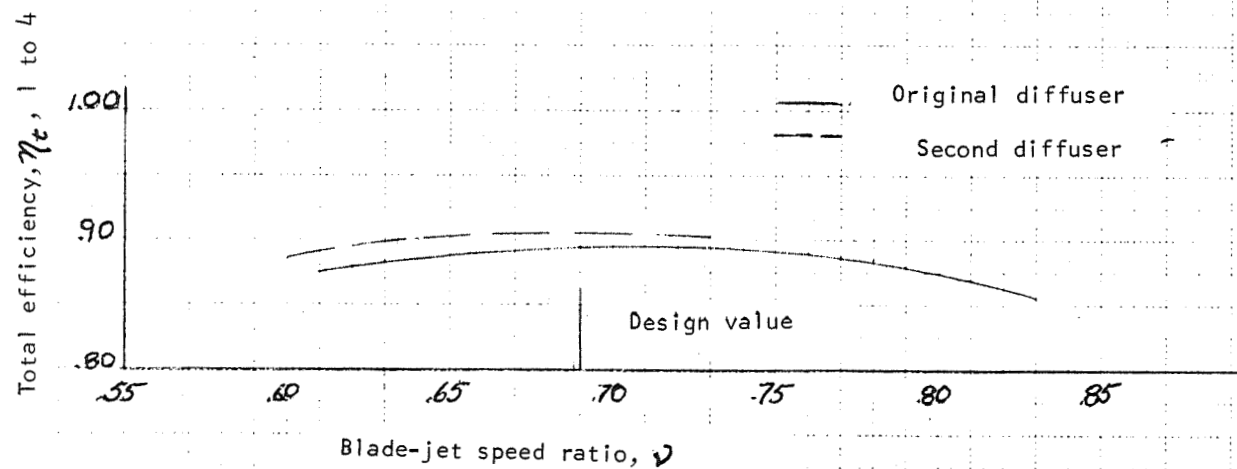


Figure 9 - Variation of total efficiency with blade-jet speed ratio for the two diffusers investigated.

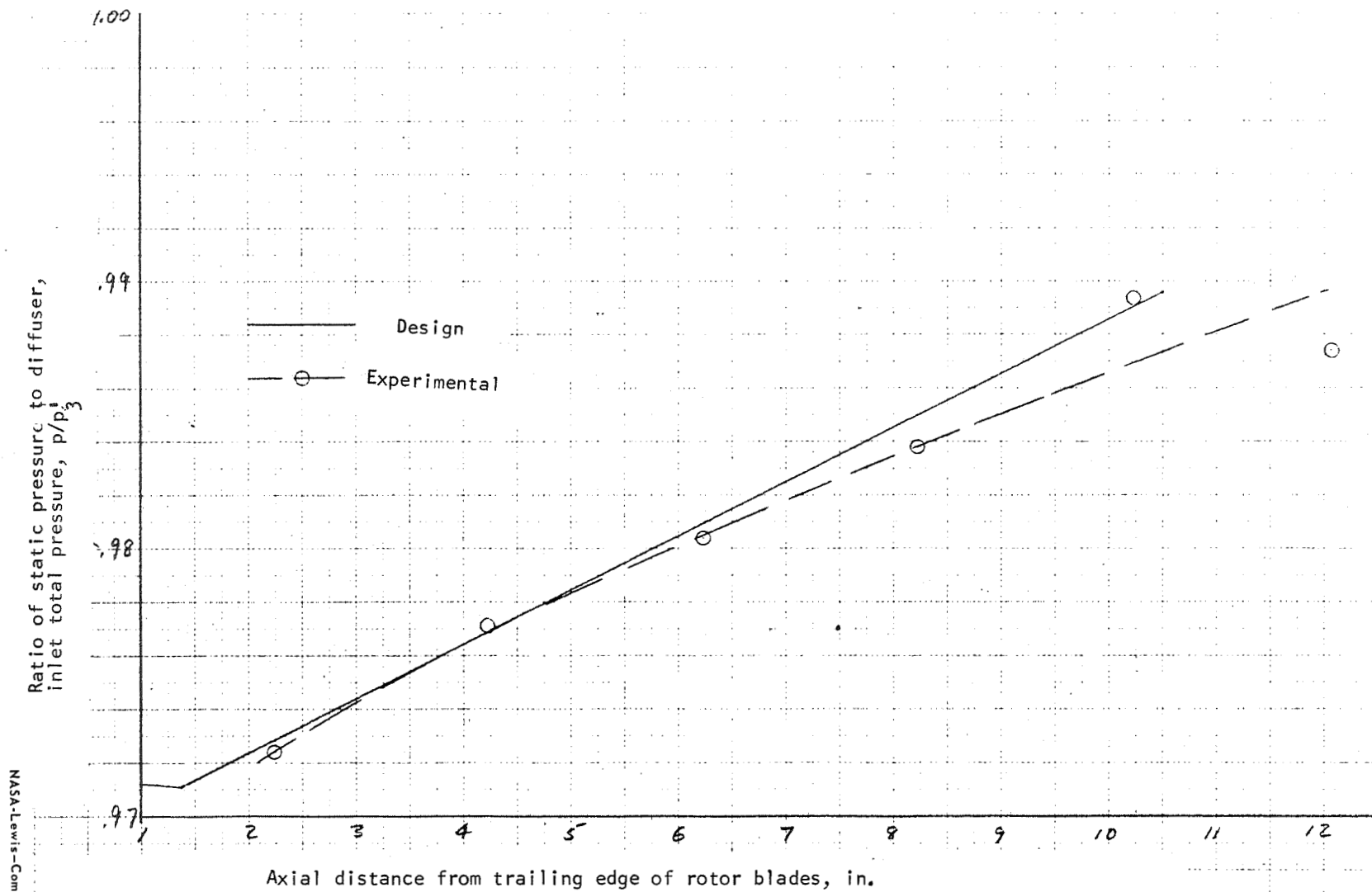


Figure 10 - Static pressures in second diffuser.